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Study of the cooling of a metal plate at high temperature by water spraying

Laurine CAPPON¹; Rabah MEHADDI¹; Tarek BEJI² and Zoubir ACEM¹

¹Université de Lorraine, CNRS, LEMTA, F-54000, Nancy, France

² Department of Structural Engineering and Building Materials, Ghent University, Tech Lane Ghent Science Park, Campus Ardoyen, Technologiepark-Zwijnaarde 60, B-9052 Ghent, Belgium.

E-mail: zoubir.acem@univ-lorraine.fr

Abstract. Experiments to cool a steel plate at high temperature by spraying water droplets were carried out using four different nozzles (TG-SS1, TG-SS3, TG-SS5, TG-SS10), in order to assess their effectiveness. Two plate thicknesses were studied. The surface temperature was measured at several points positioned diagonally on the plate: in the centre, near the edge and in between, using K-type thermocouple wires welded directly to the surface of the plate with a separate contact. This technique ensures accurate temperature measurement throughout the cooling process. The spray nozzles have been characterised to obtain an average droplet velocity V_{mean} and a Sauter mean diameter (D_{32}). Prior to spraying, the plate was heated by a radiant panel to reach steady-state temperatures, ranging from 120°C to 650°C. During cooling, the plate continued to be heated by the radiant panel. Heat flux (\dot{Q}) and heat transfer coefficient (HTC) were calculated from the temperature data. It was found that high levels of critical heat flux ($MW.m^{-2}$) were achieved in the centre of the plate, ranging from 0.86 $MW.m^{-2}$ for the TG-SS1 nozzle to 3.58 $MW.m^{-2}$ for the TG-SS10 nozzle. In the same time, the convective heat transfer coefficient also reached high values ranging respectively from 5.5 $kW.m^{-2}.K^{-1}$ (TG-SS1 nozzle) to 27 $kW.m^{-2}.K^{-1}$ (TG-SS10 nozzle). It was concluded that the mass flux of water is the parameter with the greatest influence on cooling efficiency.

1. Introduction

Water droplet spray cooling is a widespread method of heat removal used in a variety of industries, including electronics [1], metalworking and fire safety. Spray cooling involves a stream of small droplets hitting a heated surface to remove a large amount of heat. This technique takes advantage of the latent heat of evaporation, in addition to the considerable convection effects due to the impact of the droplets on the surface. Spray cooling is distinguished by its efficiency in using small quantities of fluid for high heat transfer. In the field of fire safety, water droplet spray cooling is essential for its ability to effectively absorb heat and rapidly reduce the temperature of flames and surrounding materials [2]. This method not only suppresses [3] and extinguishes fire [4], but also protects heat-sensitive equipment and structures [5]. The evaporation of water, by absorbing a significant amount of heat, helps to prevent thermal and structural damage. Efficient cooling of combustible surfaces in the early stages of a fire prevents their thermal decomposition and pyrolysis, thus preventing the spread on flames and the subsequent worsening of the fire. Moreover in many regions of the globe, water is an abundant and inexpensive resource, making



these spraying systems particularly suitable and cost-effective for a wide range of fire-fighting applications. They are therefore the preferred choice for effectively protecting materials and equipment against the destructive effects of fire. Although water-based fire extinguishing has proved effective, its modelling is underdeveloped. In fact, there is very little reliable and complete experimental data that can be used to describe well the fluid dynamics and therefore simulate the cooling of a surface as well as possible. The efficiency of water spray cooling relies on several heat transfer mechanisms. However, predicting these mechanisms is complex due to the multitude of parameters that influence the process. Indeed, elements such as droplet diameter, which influences the heat exchange surface, the velocity of the droplets V , which affects their impact, and the mass flux G , which determines the quantity of water delivered to the plate, play an important role. In addition, the modelling of spray cooling is even more complex since each type of nozzle has its own shape and its own size and velocity distributions. The distance between the nozzle and the surface to be cooled is also determinant, as it influences the interaction time and dispersion of the droplets before they reach the target. The initial temperature of the plate is also a parameter that can influence cooling efficiency. It is therefore interesting to evaluate the relative influence of each parameter on the heat transfer between the plate and the water droplets. The interaction between a spray and a hot plate has been the subject of numerous studies. G. Liang and I. Mudawar have published a comprehensive two-part review that synthesises and compares research on spray cooling [6, 7]. The study focuses on the four main fluid regimes of the boiling curve: the film boiling regime, the transition boiling regime, the nucleate boiling regime and the natural convection regime. Each of these regimes is studied individually. In all the studies carried out, there were contradictions regarding the influence of the parameters. The objectives of this study are to gain a better understanding of the mechanisms that control the cooling phenomena and to obtain data to design a reliable numerical model. To this end, a study of the hydrodynamic parameters of the spray (mean droplet velocity, Sauter diameter, local volumetric flux) will be carried out, as well as a quantitative study of the cooling efficiency of the metal plate. To do this, the temperature of the plate is monitored using K-type thermocouples placed at three different points, in order to obtain a representative measurement. Several nozzles are used to obtain different spray patterns, droplet sizes and speeds. The height of the spray will also be analysed, as well as the influence of the initial plate temperature. In this way, the change in temperature will be used to determine the heat flow and therefore the Critical Heat Flux (CHF) and the convective Heat Transfer Coefficient (HTC).

2. Experimental methods

The experimental set-up dedicated to the study of cooling includes an S235 steel plate, a radiant panel and a water spray. It is shown in Fig.1. Two different steel plates were used, each with a surface area of 1 m^2 . The thickness of the first plate is 3.1 mm and that of the second is 4.1 mm. These plates are coated with a matt black paint that is resistant to high temperatures, thus optimising radiation absorption. During the tests, the plate is placed horizontally. The "cooled side" is the side that receives the water spray and the "heated side" is the side facing the radiant panel. Six K-type thermocouples are installed along a diagonal of the plate (three on each side), as can be seen in Fig.2. The plate temperature is then monitored at three different positions: in the centre of the plate (thermocouple "1"), in the mid-distance on the diagonal between the centre and the edge (thermocouple "3") and in the edge (thermocouple "5"). The chromel and alumel wires, approximately 5 mm apart from each other, are welded directly to the surface of the plate. The steel plate itself provides the electrical contact between the wires, enabling an average surface temperature to be measured. This separate point contact configuration minimises interference such as direct exposure of the thermocouples to radiation or the effects of a thermocouple junction being immersed in a drop of water. As a result, these thermocouples provide an accurate, local measurement of surface temperature at various points.

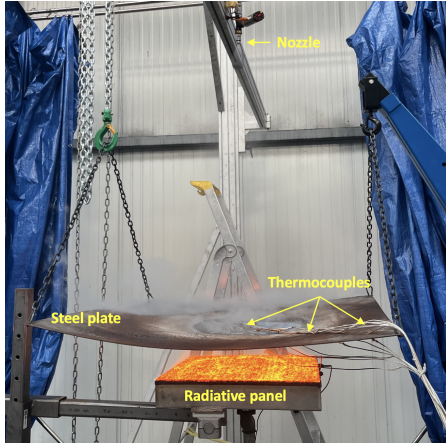


Figure 1: Experimental setup



Figure 2: Thermocouples positions

Table 1: Nozzle characteristics

	Orifice diameter (mm)	Flow (l/min)	Pressure (bar)
TG-SS1	0.914	1	
TG-SS3	1.575	3	6
TG-SS5	2.08	5	
TG-SS10	2.77	10	

The S235 steel plate is heated with a propane-fuelled radiant panel, positioned at a constant distance of 20 cm below the plate, which is placed horizontally. The panel, measuring 50 x 50 cm, delivers a total power of 50 kW, corresponding to a heat flux of 200 kW/m^2 . Prior to the experiment, the total and radiative heat fluxes of the panel were measured to characterise its performance. For example, at a gas flow rate of 1.4 g/s, the total heat flux is 39 kW/m^2 and the radiative flux is 28 kW/m^2 , leading to an equilibrium temperature at the centre of the plate of $670 \text{ }^\circ\text{C}$. So to adjust the initial temperature of the metal plate before the spraying is activated, the gas flow of the radiant panel is modified. The typical used gas flow rates were 0.25, 0.5, 1 and 1.4 g/s. Four nozzles from Spraying Systems[®] were tested: the TG-SS1 nozzle, the TG-SS3 nozzle, the TG-SS5 nozzle and the TG-SS10 nozzle. However, only the TG-SS5 and TG-SS10 nozzles were characterised. The orifice diameter, flow rate and operating pressure of each nozzle are listed in the Tab.1. The nozzles were characterised with the SpraySpy[®] laser from AOM systems. It uses the temporal phase shift method [8] to determine the spray velocity and droplet diameter distributions. The Tab.2 summarises the Sauter diameters, average velocities and Weber number for each nozzle at each distance. Moreover, the distribution of the mass flux of the sprays have been also determined using the bucket method.

3. Results and discussion

To assess cooling efficiency, it is necessary to determine the critical heat flux (CHF) and the heat transfer coefficient (HTC). CHF is the maximum value of heat flux (\dot{Q}) that a heated surface can transfer to a fluid before a change in boiling regime occurs. The HTC is a direct measure of the efficiency with which heat is transferred between two systems. Following the same approach

Table 2: Summary of mean velocity, Sauter diameter and Weber number for each configuration

Nozzle	Distance	V_{mean} (m/s)	D_{32} (μm)	Weber Number
TG-SS5	5 cm	34.9	256	4271
	50 cm	32.9	410	6079
	1 m	19.7	411	2185
TG-SS10	5 cm	38.5	283	5746
	50 cm	28.4	355	3922
	1 m	22.8	407	2898

as Cheng *et al.* [9], assuming one-dimensional heat transfer, the heat flux (\dot{Q}) can be calculated with Eq.1.

$$\dot{Q} = \lambda \frac{\Delta T}{e} \quad (1)$$

where λ is the thermal conductivity, e is the thickness of the plate and ΔT is the temperature difference between the cooled and heated sides.

The heat transfer coefficient (HTC) in a stationary state can be calculated using Eq.2

$$\text{HTC} = \frac{\dot{Q}}{\Delta T_{cs,w}} \quad (2)$$

where $\Delta T_{cs,w}$ is the temperature difference between the cooled side and the water used to cool the plate, which is at 10°C. Eq.2 describes a stationary state situation, where heat transfer is constant over time. In such a state, the system has not necessarily reached thermal equilibrium in the sense that all parts of the system are at the same temperature, but the rate of heat transfer is stable. It is assumed that the heat flux is totally transferred between the plate and the water and that there is no heat stored or released by the plate. The model therefore calculates the heat transfer coefficient (HTC) by assuming that the convective heat transfer between the surface and the fluid is proportional to the temperature difference between the solid surface and the fluid $\Delta T_{cs,w}$.

Fig. 3(a) shows the values of heat flux (\dot{Q}) (MW/m^2) and heat transfer coefficient HTC ($\text{kW}/\text{m}^2/\text{K}$) for cooling with the TG-SS10 nozzle at a distance of 1 m between the nozzle and the plate, for a gas mass flow rate of 0.5 g/s and for the central position of the plate. Fig. 3(b) shows the evolution of the temperature as a function of time under the same conditions.

The heat transfer response of a surface to spray cooling can be quantified with the help of the boiling curve, also known as the Nukiyama curve [10], shown in Fig.3 (a). This curve helps to visualise the different boiling regimes encountered during the cooling process. The boiling regimes are directly correlated to the heat transfer taking place, and depend mainly on the temperature difference between the surface and the water. The boiling curve represents the variation in heat flow from the surface to the water as a function of the superheat of the wall (wall temperature minus the saturation temperature of the liquid). There are four boiling regimes: The film boiling regime, the transition boiling regime, the nucleated boiling regime, the single-phase regime (natural convection).

In addition, Hall and Mudawar found a correlation between CHF and the mass flux G , given in Eq. 3.

$$\text{CHF} = 5.47 \cdot 10^5 \cdot G^{0.604} \cdot d_{32} \quad (3)$$

Ciofalo *et al.* also identified two correlations between CHF (Critical Heat Flux) and mass flux G as follows

$$\text{CHF} = 2.57 \cdot 10^6 \cdot G \quad (4)$$

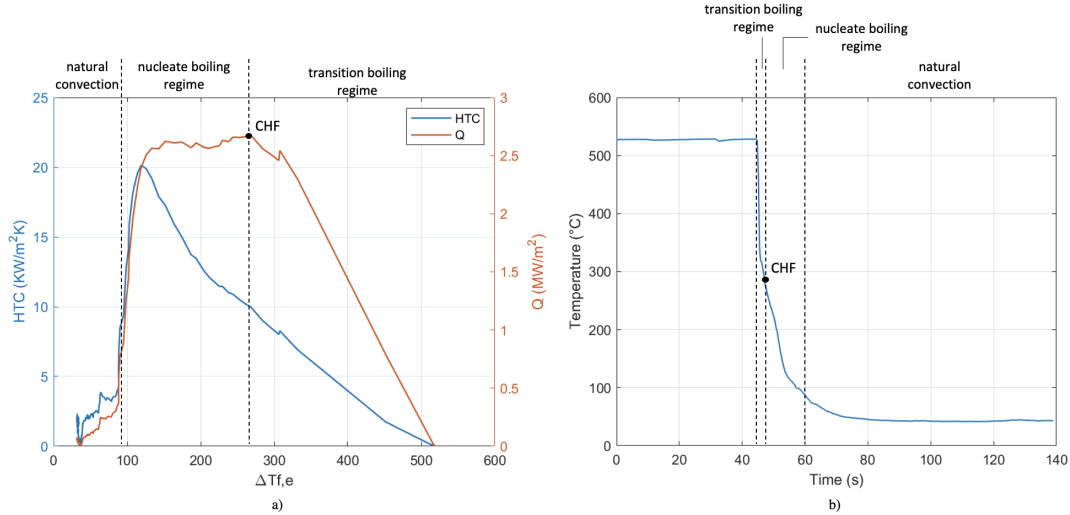


Figure 3: a) Boiling curve and b) Quenching curve for TG-SS10 nozzle

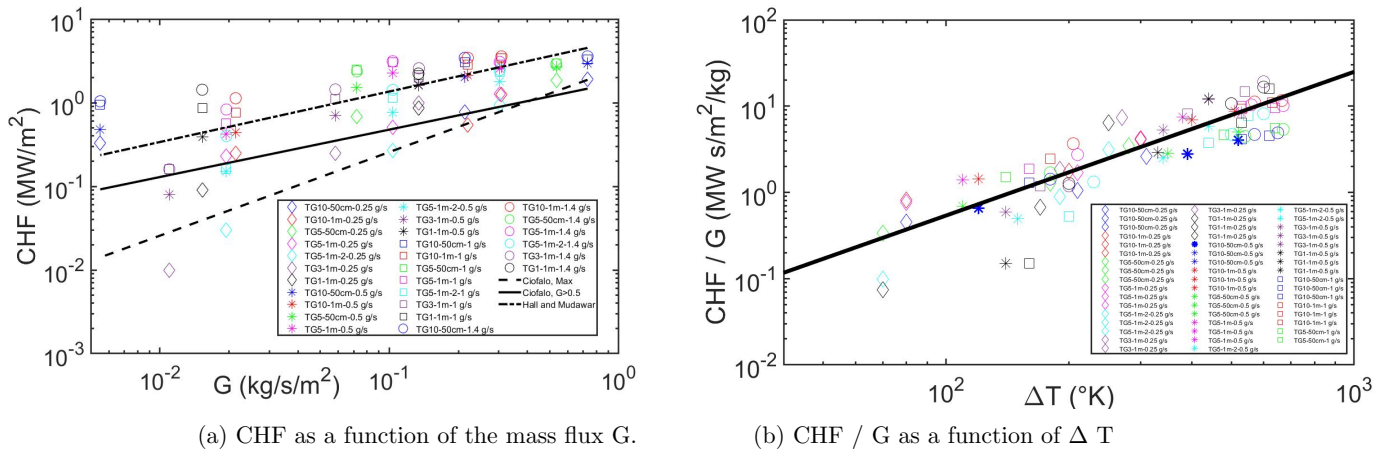


Figure 4: Critical heat flux as a function of the mass flux G . Results compared with three correlations

$$CHF = 1.759 \cdot 10^6 \cdot G^{0.567} \quad (5)$$

Eq.4 describes the model where the heat flux would be the maximum that could be removed and Eq.5 applies to cases where $G > 0.5 \text{ kg/m}^2/\text{s}$. At this level of flux, they observed a noticeable change in the slope of the correlation curve. Figure 4a shows the variation of the critical heat flux (CHF) as function of the local water mass flux G with curves that follows the slopes of G given by equations 3, 4 and 5. As can be seen, it seems that $CHF \propto G$. Moreover, figure 4b shows CHF/G as a function of the temperature difference ΔT . In addition, a straight line with a slope of $\Delta T^{5/3}$ has been added as the best fit of the present experimental investigation. Therefore, one can conclude that

$$CHF \propto G \Delta T^{5/3} \quad (6)$$

4. Conclusions

Spray cooling experiments on a hot plate were carried out to evaluate the cooling efficiency of four nozzles. Heat fluxes \dot{Q} and heat transfer coefficients HTC were determined at different positions on the plate. Due to space limitations, it was not possible to present all the results, but the main conclusions drawn from this study are:

- (i) Droplet diameter increases with distance. For the TG-SS5 nozzle, the diameter increases from $256\ \mu\text{m}$ to $411\ \mu\text{m}$ when the distance from the nozzle increases from 5 cm to 1 m. At the same time, velocity decreased: for the TG-SS5, it fell from 34.9 m/s to 19.7 m/s. For the TG-SS10 and over the same distances, the diameter varies from $283\ \mu\text{m}$ to $407\ \mu\text{m}$ and the speed drops from 38.5 m/s to 22.8 m/s.
- (ii) CHF values range from $0.86\ \text{MW}\cdot\text{m}^{-2}$ to $3.58\ \text{MW}\cdot\text{m}^{-2}$, and are mainly influenced by the mass flux, but also by the number of Weber and the type of nozzle used.
- (iii) HTC maximum values range from $5.5\ \text{kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ to $27\ \text{kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$, which is very far from the default value ($300\ \text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$) between water and solid surfaces used in FDS.
- (iv) A smaller droplet diameter favours CHF, as does a higher mass flow.
- (v) The TG-SS10 nozzle has better CHF than the TG-SS5 nozzle, but the difference between these CHF is not significant in relation to the volume of water it uses, which is twice as much.

This study offers a number of perspectives for improving our understanding of cooling phenomena. For example, the plate could be heated with a $1\ \text{m}^2$ radiant panel to even out the initial temperature of the plate before spraying. It would also be useful to test the cooling at a distance of 50 cm on a thicker plate, in order to supplement the data in figure 4. Exploring other plate configurations, such as a vertical orientation, or testing greater plate thicknesses, could provide additional information.

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